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Three Decades of Modern Practice in Screw Compressors

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ABSTRACT

The mathematical modelling of screw compressor processes and its implementation in their design began 30 years ago with the publication of several pioneering papers on this topic, mainly at Purdue Compressor Conferences. This led to the gradual introduction of computer aided design, which, in turn, resulted in huge improvements in these machines, especially in oil-flooded air compressors, where the market is very competitive. A review of progress in such methods is presented in this paper together with their application in successful compressor designs. As a result of their introduction, even small details are now considered significant in efforts to improve performance and reduce costs. Despite this, there are still possibilities to introduce new methods and procedures for improved rotor profiles, design optimisation for each specified duty and specialized compressor design, all of which can lead to a better product and new areas of application. This paper gives a review of methods and procedures which lead to modern screw compressor practice.

1. INTRODUCTION

During the past thirty years, for many applications, traditional reciprocating compressors have been displaced by those of the twin screw type. The main reasons for this are the improved rotor profiles, which have reduced internal leakage, and machine tools, which can manufacture the most complex shapes, to tolerances of 3 micrometers, at an acceptable cost. Although, advances have been made in analytical procedures, which are gradually being adopted by designers to predict compressor performance more reliably, these improved methods of analysis can create, as yet unrealised, opportunities for further improving the performance and reducing the cost of screw machines.

Rotor profile enhancement is still a means of further improving screw compressors and rational procedures are now being developed both to replace earlier shapes and also to vary the proportions of any selected profile to obtain the best result for the application for which the compressor is required. In addition, improved modelling of flow patterns within the machine can lead to better porting design. Also, more accurate determination of bearing loads and how they fluctuate enable better choices of bearings to be made. Finally, if rotor and casing distortion, as a result of temperature and pressure changes within the compressor, can be estimated reliably, machining procedures can be devised to minimise their adverse effects.

Up to date modelling and analytical procedures, now being developed to address all these possibilities, are reviewed here together with examples of how their utilisation has led to improved designs and new applications.

1.1 Background

Screw compressors in normal commercial usage today have main rotors whose outer diameters vary between 75 mm and 620 mm. These deliver between 0.6 m^3/min and 600 m^3/min of compressed gas. The normal pressure ratios attained in a single stage are 3.5:1 for dry compressors and up to 15:1 for oil flooded machines. Normal stage pressure differences are up to 15 bars, but maximum values sometimes exceed 40 bars. Typically, for oil flooded air compression applications, the volumetric efficiency of these machines now exceeds 90 % while specific power inputs, which are both size and performance dependent, have been reduced to values which were regarded as unattainable only a few years ago.

In order to operate effectively a sealing line must be formed between the rotors and between the rotors and the casing. A small gap, the so called "blowhole", occurring between the cusp of the casing and the rotors, extends along the length of the casing to form a path through which the gas being compressed leaks. The aim is to select a rotor profile which maximises the flow while minimising the blowhole area, the sealing line length and the contact forces between the male and female rotors. Although this principle requirement for screw compressors has been known for over 100 years, it is only since the development of the SRM "A" profile by *Shibbie, 1973*, which met these criteria far better than any of its predecessors, that screw compressors began to be commercially viable.

1.2 Publications

Despite the rapid growth in screw compressor usage, the scientific basis of their design is still limited. Several screw compressor textbooks were published in Russia in the early nineteen sixties. *Sakun 1960* gives a full analysis on rotor profiles based on the envelope method. *Andreev, 1961* gives more information on manufacturing and tooling for screw compressor rotors, while *Golovnitsov, 1964* introduced contemporary procedures in a more general book on rotary compressors. *Amosov et al 1977* in his handbook made a contribution by giving a reproducible presentation of how to generate the SRM asymmetric profile, the Lysholm profile, as well as a presentation of the early Russian SKBK profile. In two German textbooks, *Rinder, 1979* used a profile generation method based on gear theory to reconstruct the SRM asymmetric profile while *Konka, 1988* published some detailed engineering aspects of screw compressors. There are two-three textbooks on screw compressors published in English, *O'Neil, 1993* and *Arbon, 1994*. More recently, *Xing, 2000* published a comprehensive textbook on screw compressors in Chinese. More recently, the authors published two books on screw compressors, *Stosic et al, 2005 and 2006.* A few compressor manufacturers' handbooks on screw compressors and a number of brochures give useful information, but they are either classified or of a very limited accessibility.

Literally thousands of patents have been awarded for screw compressors during the past thirty years and SRM alone has more than 1000. They cover various aspects of screw compressor design but are mainly for rotor profiles. SRM patents *Nilson, 1952* for the symmetric, *Shibbie, 1979*, for the asymmetric, *Astberg, 1982* for the "D" and *Ohman, 1999* for the "G" profiles are examples of state of the art screw compressor profile generation. Other successful profile patents are those of *Bammert, 1979*, *Hough et al, 1984*, *Edstroem, 1974*, *Kasuya et al, 1983* and *Bowman, 1983*. More recently, several successful patents such as *Lee, 1988* and *Chia-Hsing C, 1995* have been granted to relatively small companies. A fresh approach to profile generation based on the use of a rack for the primary curves was published in *Rinder, 1987* and *Stosic, 1996*. Although all patented profiles were generated by well defined procedures, so little was published about the methods on which they were based that it was difficult to reproduce them.

Most of the known characteristics of screw compressors, such as oil flooding, the shaping of the suction and discharge ports to follow the rotor tip helices, axial force compensation, unloading, the slide valve and the economizer port, were also patented, mainly by SRM. However other companies were equally keen to file patents. It appears that, in the field of screw compressors, patent experts are as important as engineers.

Three conferences dominate the screw compressor area, Purdue Compressor Engineering Conference, Dortmund VDI Tagung 'Schraubenmaschinen' and the IMechE Conference on Compressors and their Systems, London. Modern screw compressor practice started with calculation of the compressor process, based on the solution of differential equations derived from the conservation of mass and energy and temperature and pressure relationships derived from equations of state. This was strongly supported in Purdue publications, early examples of which are *Benson et al, 1972, MacLaren et al, 1974* and *Prakash et al, 1974* in reciprocating compressors. In screw compressors, *Fujiwara et al, 1974* and *1984*, *Fukazawa et al, 1980*, as well as *Sangfors, 1982* and *1984*, *Bain et al, 1982*, then *Singh et al, 1984* and *1990, Dagang et al, 1986* and later *Edstroem, 1992* all made contributions. *Stosic et al* introduced a solution of the energy and mass differential equations in their primitive form in *1988*. The Dortmund proceedings contain some interesting screw compressor papers. *Rinder, 1984* presents a rack method of rotor profile generation, based on classical gearing procedure, which is fully reproducible. *Sauls, 1998* gives more details on profiling and manufacturing control. *Kauder and Harling, 1994* showed a typical example of a successful university research applied to solve real engineering problems. *Edstroem 1989* published an interesting paper at the IMechE Conference, which was followed by other papers, such as those of *Venumadhav et al and McCreath et al, 2001* and more recently *Stosic et al, 2005, 2007* and *Delash et al, 2009.*

There are surprisingly few journal publications on screw compressors in the technical literature. An early exceptions was that of *Margolis 1978*. Others followed in the Journal of the International Institution of Refrigeration, such as those of *Stosic et al 1992* and *Fujiwara and Osada, 1995*, and then in the IMechE proceedings by *Tang et al, 1994*, and *Fleming et al 1999* and *Hanjalic and Stosic, 1997* in the ASME Journal of Fluids Engineering. All of them either introduced or contributed to a mathematical model based on solution of the differential equations of mass and internal energy in terms of their primitive variables, which is indeed the backbone of the modern approach to the analysis of screw compressors. This was followed by *Stosic 1998* and more recently, *Stosic 2004*, as well as *Nouri et al, 2007* in various engineering journals. This led to the presentation of more information on screw compressors in journals than in all previous years together.

2. SCREW COMPRESSOR DEVELOPMENTS

Since tight clearances are achievable nowadays, internal compressor leakage rates have become small. Hence, further improvements in screw compressor design are possible only by the introduction of more refined analytical principles.

The main requirement is to improve the rotor profiles so that the internal flow area through the compressor is maximised while the leakage path is minimised and internal friction due to relative motion between the contacting rotor surfaces is made as small as possible. This is achieved through several steps described in this chapter.

2.1 Rotor Configuration

Increasing the number of rotor lobes enables the same built-in volume ratio to be attained with larger discharge ports. Larger discharge ports decrease the discharge velocity and therefore reduce the discharge pressure losses, thereby increasing the compressor overall efficiency. Hence screw compressors tend to be built with more lobes than the traditional 4-6 combination and 5-6 and 6-7 configurations are becoming increasingly popular. Also, the greater the number of lobes, the smaller the pressure difference between the two neighbouring working chambers. Thus, interlobe leakage losses are reduced. Furthermore, more lobes combined with a large wrap angle ensure multiple rotor contacts which reduce vibrations and thus minimize noise. However, more lobes usually mean less rotor throughput, which implies that rotors with more lobes are somewhat larger than their counterparts with fewer lobes. Also the leakage to delivery ratio is worse with more rotor lobes. Therefore, such compressors are less efficient. Additionally, more lobes increase the manufacturing cost.

2.2 Rotor Sealing Line Length and Blow-Hole Area

Since screw compressors tend to rotate relatively slowly, rotor profiles must have the smallest possible blow-hole area if leakages are to be minimised. However, reduction of the blow-hole area is associated with increase in the sealing line length. It is therefore necessary to find the optimum profile shape which minimises the sum of both the blow-hole and sealing line leakage areas.

2.3 Rotor Proportions

A general feature of screw compressors is that the pressure difference through them causes high rotor loads and this is especially the case for low temperature refrigeration compressors, where these are large. Therefore, to maintain their rigidity and minimise deflection, rotor profiles usually have a relatively small male rotor addendum in order to increase the female root diameter. This sometimes leads to very shallow and clumsy rotors. An alternative possibility is to increase the female rotor lobe thickness. This greatly increases its moment of inertia and thereby reduces the rotor deflection more effectively.

2.4 Rotor Wrap Angle

Increasing the rotor wrap angle is generally associated with reducing the interlobe sealing line and hence, with reduced leakage between the rotors. Contemporary trends in screw compressor design are therefore towards larger wrap angles. However, on occasion, this has led to exceeding the limiting values and thereby reducing the compressor displacement.

2.5 Compressor Bearings

In some compressor designs, multiple cylinder roller bearings or multipoint ball bearings are located at the high pressure end of the rotors to withstand the large radial forces reliably over a long operating life, for example, *Meyers, 1997*. Frequently, two bearings are also employed for axial loads. Since only one axial bearing actually takes the load, the role of the other is mainly to prevent rotor bounce in the axial direction.

2.6 Rotor Clearance Distribution and Contact on the Lobe Flat Side

Oil flooded compressors have direct contact between their rotors. In well designed rotors, the clearance distribution will be set so that this is first made along their, so called, contact bands, which are positioned close to the rotor pitch circles. Since the relative motion between the contacting lobes in this region is almost pure rolling, the danger of their seizing, as a result of sliding contact, is thereby minimised. The traditional approach is to maintain a high, so called, positive gate rotor torque, which ensures round flank contact, *Edstroem, 1992*. What is not widely appreciated is that there are significant advantages to be gained by maintaining a negative gate rotor torque to ensure that contact, when it occurs, will be on the flat face. The reason for this can be understood by examination of the sealing line lengths that for the flat flank is much longer than that of the round flank. Thus, minimising the clearance on the flat flank will reduce the interlobe leakage more than minimising the round flank clearance. Also, negative gate torque is achieved by making the gate rotor lobes thicker and the main rotor lobes correspondingly thinner. The displacement is thereby increased. Thus both these effects lead to higher compressor flows and efficiencies.

2.7 Thermal Expansion of the Rotors and Housing

Although the temperature range over which screw compressors operate is not large, the effects of thermal expansion are highly significant if the small clearances required between the rotors and between the rotors and the housing are to be maintained under working conditions. Thus, the rotor clearances obtained under manufacturing conditions must be estimated while taking account of thermal distortion that will occur when the compressor reaches its operating temperature and pressure and calculation must allow for unequal expansion of the rotors in different coordinate directions. An example of this is given in Fig. 1, where the left diagram shows the estimated clearance distribution when the rotors are cold, while, the centre and right diagrams show the clearances after the rotors reach their working temperatures. Additional information about screw compressor clearance management and other means of improving efficiency may be found in *Stosic et al, 2004*.

2.8 Displacement of Bearing Centres

One additional design aspect, which though important, is not widely appreciated is that the pressure loads will tend to push the rotors apart from their design position in the casing, as a result of the clearances within the bearings. If these are not taken fully into account, the resulting displacement will cause contact between the rotor tips and the casing, when the rotor clearances are small and the pressure loads are high. To counter this, the bearing centre distance must be smaller than that of the rotor housing bores. To maintain the rotor interlobe clearance as small as possible, the bearing centre distance must be even further reduced.

Also, if the bearing centres are set to be the same as those of the rotors, the clearance between the rotors and housing will be smaller at the low pressure side of the rotors and larger at the high pressure side. Since leakage is caused by the pressure difference, this displacement creates the least favourable rotor position for efficient compressor operation. The bearing centre distances must therefore be arranged to maintain a uniform clearance between the rotors and the housing.

2.9 Optimisaton of the compression process

Analysis of compressor behaviour shows that there are conflicting requirements for desirable machine characteristics. This implies that only simultaneous optimisation of all the variables involved in the design process will lead to the best possible compressor performance. A full multivariable optimisation of screw compressor geometry and operating conditions should be performed to establish the most efficient compressor design for a given duty. This can be achieved by the use of a computer software package, based on a Box constraint simplex method, which provides the general specification of the rotor and compressor characteristics in terms of several key parameters and which can generate various rotor and compressor shapes. For example, see *Stosic et al, 2003*.

3. EXAMPLES OF COMPRESSOR DESIGNS

Many screw compressor manufacturers have followed the more up to date practices described. Eight examples of these are taken from the recent publications, in which some or all of the features described in the previous chapter have been taken into account. In addition, three projects are presented in this paper for the first time.

3.1 Rotor Retrofit for Efficient Screw Compressors

Since the market for oil-flooded screw air and refrigeration compressors is highly competitive, new designs are continually being introduced which are more efficient and cost effective than their predecessors. However, because of the high cost of development of new machines, manufacturers seek to maintain their existing designs for as long as possible. Closer study of many of the older designs has shown that in the majority of cases, all that is required to bring them up to date is to change the rotor profile to one of a more recent type. An example of this is given by *Stosic et al, 2000,* which describes the retrofit of new rotors into an existing family of oil-flooded compressors instead of A rotors. The old and new rotors are compared in Fig. 3.

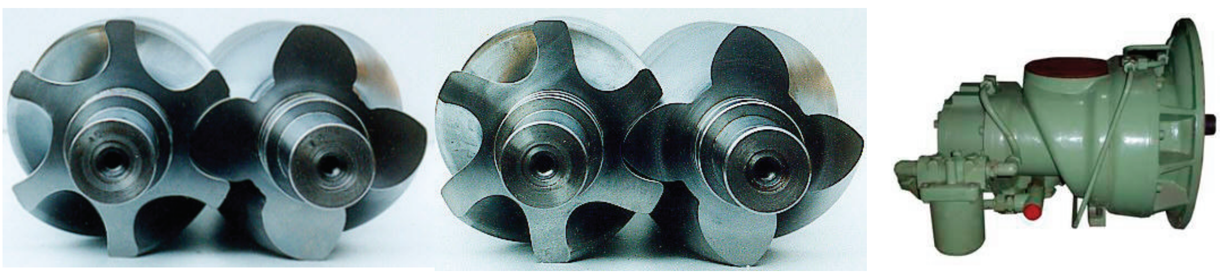

Fig. 1 Rotor retrofit in oil-flooded screw compressors, old rotors left, new rotors, right, Stosic et al, 2000

More recently, SRM 'A' rotors were replaced by new rotors in classic open refrigeration compressors, as described by *Zhang et al, 2006*. A similar exercise where the symmetric externally synchronized 4/6 rotors in a semihermetic refrigeration compressor were replaced by new rotors in direct contact, was described by *Delash et al, 2009*.

3.2 Screw Compressor for Delivery of Dry Air

McCreath et al, 2001 published a paper which describes two high efficiency oil-free screw compressors designed for dry air delivery. Their design is based on rack generated 3/5 rotor profiles, shown in Fig. 2.

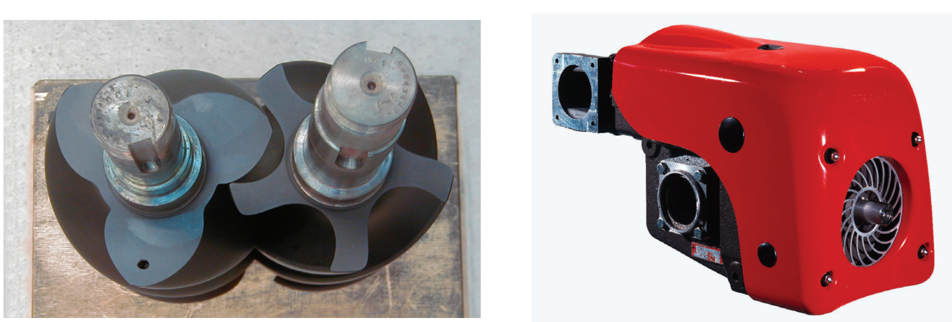

Fig. 2 Rotor profile for delivery of dry air, McCreath et al, 2001

3.2 Design of Oil-Flooded Air Compressors

The design of a family of efficient oil-flooded twin screw air compressors is described by *Venumadhav et al, 2001*. Rack generated rotors with a 4/5 configuration were applied to 5 screw compressors of 73, 102, 159, 225 and 284 mm rotor diameter, respectively, to cover air delivery of 0.6 to 60 m^3/min at delivery pressures between 5 and 13 bar. The compressor family is being gradually introduced by manufacturing prototypes, pre-production compressors and finally, production units. The compressor prototypes tests showed that the volumetric and adiabatic efficiencies of the prototypes were high when compared to published data on the best compressors currently manufactured.

Fig. 3 Two-stage oil flooded screw compressor plant

Fig. 4 Oil flooded air end

The principles of modern screw compressor practice described in the previous chapter were used in design of a family of two-stage oil flooded screw compressors, Fig. 3. The measurements preformed on the family confirmed the highest efficiency ever reported in the open literature.

Later *Stosic et al, 2006* published a design of an oil flooded compressor which exhibited the highest efficiency of a single stage oil flooded screw compressor. More recently, the principles of modern compressor practice were applied to an air screw compressor presented in Fig. 4.

3.3 Design of Refrigeration Compressors

Early works in refrigeration compressors resulted in a design presented in Fig. 5, *Zhang et al* 2006, later *Broglia et al 2006* published their work, Fig. 6 and modern practice is applied for efficiency improvements in the refrigeration compressors presented in Fig. 7.

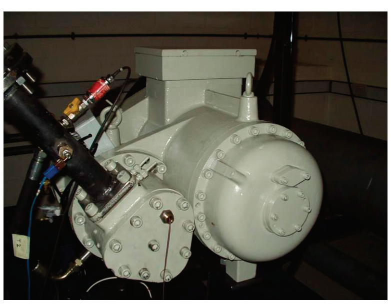

Fig. 5 Refrigeration compressor, Zhang et al, 2006

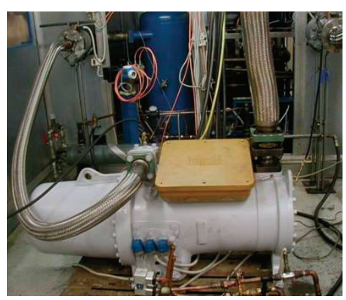

Fig. 6 Refrigeration compressor, Broglia, 2007

Fig. 7 Refrigeration compressor

5. CONCLUSIONS

Although the screw compressor is now a well developed product, greater involvement of engineering science in the form of computer modelling and mathematical analysis at the design stage, makes further improvements in efficiency and reduction in size and cost possible. Also, advances in bearing technology and lubrication, must continually be included to obtain the best results.

4. ACKNOWLEDGEMENT

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